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BUCKLING IN CIRCULAR PLASTIC CYLINDERS.

JOHN J. SEEBERGER

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BUCKLING IN CIRCULAR PLASTIC CYLINDERS

by

John J. Seeberger
Lieutenant, United States Navy
B. S. , United States Naval Academy, 1958

Submitted in partial fulfillment
for the degree of

MASTERS OF SCIENCE IN AERONAUTICAL ENGINEERING

from the

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ABSTRACT

The objective of research in the field of axially compressed cylinders is the establishment of sound design principles. Only through additional analytical and experimental effort can this goal be met. The most critical need at the present is believed to be for additional tests, which, unlike most previous tests, will be designed specifically to provide general information about the buckling process. This investigation adds to the general study on the buckling of thin-walled cylindrical shells fabricated from the polyester film, Mylar. It was concluded that the number of circumferential buckles, which controls the postbuckling behavior of cylinders, is dependent on the geometric parameters, R/t and L/R . Such dependence is approximated by de Neufville's solution:

$$N = 2.2 \left[\frac{R/t}{L/R} \right]^{1/4} .$$

It is also concluded, for the range of the R/t and L/R parameters considered in this investigation, that the ratio of the buckling stress is: (1) independent of the parameter L/R , and (2) dependent on the parameter R/t , with increasing values of the ratio R/t causing a decrease in σ/σ_{cl} .

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LIST OF SYMBOLS

σ	buckling load
σ_{cl}	classical buckling load
N	number of circumferential buckles
R	radius of cylindrical shell
L	effective length of cylinder
t	wall thickness of cylinder
E	Young's Modulus of material

1. Introduction

For over thirty years the serious discrepancy between theoretical and experimental results for the buckling of axially compressed cylindrical shells has been recognized, but the phenomenon has had little understanding. Figures 1 and 2 are block diagrams depicting the chronology on the investigation of buckling stress and buckling mode of thin cylindrical shells. In 1932 Flügge calculated the theoretical buckling stress for asymmetric buckling as $\sigma_{c1} = 0.6 Et/R$. Experimental investigations found that the ratio of the actual buckling stress to the classical buckling stress was quite low with great scatter in the test results. The study of Karman and Tsien in 1941 presented a possible explanation of the scatter in the test results and the reduction in the buckling load [6]. By the use of the theory of finite deflections, they managed to show that the shell stability is quite weak at load levels considerably below the classical buckling load. The essential result of their investigation was the statement that the axially compressed cylinder, after having reached the critical buckling load, must snap through into another state of equilibrium which is connected with a considerably smaller axial load [9]. Their theory was slowly refined in the following years. However, due to the assumption of an initially perfect cylinder, which was used in

these studies, the highest load carried according to the non-linear theory still is the buckling load predicted by classical theory. In 1950 Donnell and Wan suggested that the cause for the scatter and reduction of buckling loads was due to initial imperfections in the circular cylinder [4]. Their investigation showed that the actual buckling load is considerably reduced, compared with the classical buckling load, due to initial imperfections in the order of only a few tenths of the thickness of the shell wall, and that the difference between the actual and classical buckling loads depends on the amplitude of these initial imperfections [9]. However, the actual shape of the imperfections in their analysis is obscure and the exact degree of the effect for various types of imperfections is still to be determined.

Results in recent years by Almoth, Holmes and Brush [1], Babcock and Sechler [2], and Tennyson [8] show that very high values of the buckling load can be obtained when initial irregularities are carefully minimized. This indicates strongly that when cylindrical shells are sufficiently free of imperfections and axially loaded, the classical buckling load can be closely obtained and a lower buckling load is not always inevitable.

Whether a useful theoretical solution will ever be feasible remains to be seen, but even if cylindrical shells are designed by use of empirical methods, it is important that we understand what happens when the shell buckles.

The concept of non-linear buckling showed that a buckled state could be maintained at $\sigma < \sigma_{cl}$, but failed to show how the cylinder buckled without passing through the condition that $\sigma = \sigma_{cl}$. The mechanism of the transition is now the main problem in shell buckling.

Calculations which are presently made on the number of buckles formed around the circumference of an axially compressed cylinder are in variance with those observed experimentally. From published experimental data, the variation of the number of circumferential buckling lobes with the cylinder parameters R/t and L/R have been obtained. Hoff suggested that the number of buckling lobes are interrelated with the critical compressive loading [5]. The nature of this dependency poses a new problem in the understanding of the mechanism of the buckling of axially loaded circular cylinders.

It is important to be able to determine the number of buckling waves at failure, N , because it is an index of the buckling mode, which in turn controls the behavior and strength of the cylinder after buckling. The number of circumferential buckles can be determined from the linear Flügge equations and is approximately predicted by de Neufville's solution in 1965 as [7]:

$$N = 2.2 \left[\frac{R/t}{L/R} \right]^{1/4}.$$

De Neufville has done extensive investigations on Mylar

cylinders in the region for R/t greater than 800 and less than 1600. The major aim of the present paper is to investigate and validate de Neufville's analytical formulation for values of the parameter R/t less than 800.

The primary factors which have been considered in this report are the influence of L/R , R/t , and σ/σ_{cl} on the number of circumferential buckles. In this investigation over 300 tests were conducted on over 50 cylinders fabricated from the polyester film, Mylar.

The first part of the test program was designed to determine if a normal distribution of results would be obtained for nominally identical test models.

The remaining part of the test program was designed to determine the effect of varying the geometric parameters, L/R and R/t , on the number of circumferential buckles.

It is hoped that this study will stimulate the interest of others to perform additional work needed to complement and extend the range of geometric parameters considered, as the understanding of the buckling phenomena is still far from complete. It is also hoped that the net result of these investigations will bring some order to the study of cylindrical shells in axial compression.

2. Model Material

All of the cylindrical models used in the experimental program were fabricated from Type A Dupont Mylar Polyester Film. Mylar is readily available in thin sheets, of 0.00015" (.15 mil) to 0.014" (14 mils) in thickness, and it is inexpensive. Extraordinary folding endurance is a characteristic over the full range of temperature and humidities. Mylar was used because of its ability to undergo large elastic strains without excessive permanent set. This property enables the test model to be buckled several times in axial compression, at virtually the same buckling load--the so called "repeatability" phenomenon. According to the manufacturer, Mylar has very nearly isotropic mechanical properties, being made by a random calendering process. Mylar has been used recently in many tests of which in particular, Seide, Weingarten and Morgan specifically investigated how Mylar cylinders could be made and tested. They determined that the material has a proportional limit of about 6,000 psi, a Young's Modulus in the neighborhood of 700,000 psi and a Poisson's ratio of approximately 0.3 [10]. The nominal variation of the sheet thickness was found to be well within those limits specified by the manufacturer($\pm$ 0.001 inches).

The above values of Young's Modulus and Poisson's ratio were used for all calculations performed in this investigation.

3. Model Parameters

Since the objective of the test program was to test for the variation of the number of circumferential buckles and the post buckling characteristics of cylinders with R/t below R/t equal to 800, a wide range of geometric parameters was chosen for the tests. These values are shown in the Schedule of Parameters, Table I.

The thickness of the cylinders was a nominal 0.0075 inches (7.5 mils) or 0.010 inches (10 mils). The radius was varied from two to five inches and the length from 2.75 inches to 10.0 inches.

4. Test Models

All test models were thin-walled circular cylindrical shells with a glued lap joint along the generated line. The models were fabricated by wrapping Mylar sheet about a wooden cylindrical mandrel. The Mylar sheets were accurately cut into the desired axial lengths and the circumferential length of the desired cylindrical model plus one inch overlap for the bonding of the lap joint. Bonding was accomplished by using an uncured silicone rubber adhesive. To insure secure bonding, the overlap was restrained by a wooden bar with felt stripping, which was held securely against the lap joint by circumferential bands until the adhesive had cured. The cylindrical mandrel had a flat plate

attached at one end to aid in the forming of the test model squarely. Although the thin Mylar sheet is quite flexible, the use of carefully laid out patterns and wooden mandrels made it possible to fabricate test models that were dimensionally accurate and relatively free of initial wrinkles or bulges.

End clamping was accomplished by inserting the model into a circular cavity of an accurately machined end plate and filling the cavity with rubber cement. This edge condition was utilized for all the tests conducted in this investigation. Great care was taken to insure that the end plates were parallel so as to obtain stress uniformity when the load was transmitted to the test model by means of the end plates.

It has often been observed experimentally that a small internal pressure will raise the buckling load of a cylinder significantly. To obtain comparable data it is essential to maintain a uniform pressure. Ventilation holes were therefore cut in the end plates in order to equalize internal and external pressures so that no pressure buildup would occur.

The test model fabrication procedure is pictorially displayed in Figure 3. A view of the various test model sizes is shown in Figure 4.

5. Testing the Model

The test machine used in this investigation was a standard

Baldwin 60,000 pound Universal Test Machine. All of the tests conducted were performed on this machine. The test models were centered accurately relative with the testing machine. The load was applied until buckling had occurred.

In this type of investigation it is most important that the stress is evenly distributed around the circumference of the test model. The alignment of the test model with the test machine loading plate was accomplished by the use of a ball joint. This technique is the most convenient method of insuring that the load is applied parallel to the test ~~model~~ axis. The ball joint consists of two metal circular plates in each of which is cut a spherical dome, and into which a greased ball is inserted. Initially in the testing procedure the ball joint is placed at the center of the end plate where the load is applied.

The first few tests conducted on a given model were used to adjust the position of the ball joint so as to obtain as nearly as possible a symmetrical buckling pattern. Proper alignment was then judged to have been achieved, regardless of the position of the ball joint through which the load was applied.

When the test model had failed, the load was held as near as possible at the drop-off value while the buckles it had formed were measured. For 60% of the models investigated the circumference of the model was not filled completely with buckles;

the buckles that formed were measured. The wavelength was then used to extrapolate the buckles to the complete circumference.

The loading and unloading cycle was repeated several times on each test model to obtain repeatable values of the buckling load and pattern. After each unloading the ball joint was displaced away from the direction where the previous buckling pattern had been initiated. This procedure helped to insure proper centering.

A minimum of five tests was completed on each test model. The testing machine and model setup are shown in Figure 5 .

6. Results

Over 300 tests were conducted on over 50 test models to obtain sufficient data for the analysis. Room temperature for all tests was approximately 72°F, plus or minus two degrees. There was no control on the humidity which was assumed to have remained constant.

Test model failure was characterized by the snap through action of the shell wall into the buckled condition. Figure 6 is a photograph of a typical buckling pattern observed during the tests.

Bad results were catagorized into obviously bad (poor fabrication rejects-two test models), and suspected bad (outside the population of the normal pattern of results-four test models). Bad tests were probably due to: obvious imperfections of the test model, which were visually observed, or test results (probably due to imperfections)

which statistically fell outside the population of the normal pattern of results. Obvious imperfections were due to poor fabrication or wrinkling of the test model prior to buckling.

For the majority of tests the buckling initiated locally and it was necessary to extrapolate the buckle wavelength around the circumference of the model to obtain the number of circumferential buckles. This condition is attributed to a non-uniform stress pattern in the wall of the model which is caused by an off centered loading or a tilted shell. Obviously, this results in a decrease in the buckling load compared to the ideal experimental condition. In this analysis the above condition was acceptable, since as long as the models tested supplied results which fell into a homogeneous population, they were meaningful in the buckling analysis.

Results of the test program conducted are found in Tables II and III with pertinent relations graphically displayed in Figures 7 through 16.

The first series of tests, identified as the control tests, were conducted on test models of the same geometry in order to determine if results would fall into the same population. Test results are found in Table II, Summary of Data for Control Tests, with a graphical display of the results in Figures 7 and 8. The results of the control tests indicate that the frequency of distribution of the buckling load and the circumferential buckle

number, N , is normal and that the small scatter of results is about a definite meaningful value.

From Table III it is obvious that as the testing program progressed the results obtained were more consistent. This is attributed to the learning process, whereby as the testing and fabrication program progressed there was more experience to rely upon.

As shown in Figure 9, Variation of N with R/t and L/R , Theory and Experimental Results Compared, the tests confirm the primary objective of the analysis. The number of circumferential buckles for the region of R/t less than 800 is definitely determined by the geometric parameters R/t and L/R and the value of N is approximately predicted by de Neufville's theoretical expression:

$$N = 2.2 \left[\frac{R/t}{L/R} \right]^{1/4} .$$

It is noted that the test results fall very close to those theoretically predicted.

From Figures 10 through 15 it is seen that the variation of L/R alone does not change σ/σ_{cl} . Investigation of the values of σ/σ_{cl} obtained for the ratios of L/R chosen in the analysis at a given R/t ratio shows that σ/σ_{cl} is independent of the parameter L/R . This result is in agreement with a recent analysis by another investigator [9]. It is also observed

from Figures 10 through 15 that as the parameter R/t is increased in value, the ratio of σ/σ_{c1} decreases.

From Figure 16, Variation of σ/σ_{c1} with R/t , it is shown that as R/t increases, the value of σ/σ_{c1} decreases. This result is contrary to de Neufville's conclusion that σ/σ_{c1} was independent of the parameter R/t . In Figure 16 it is noted that the parameter R/t has its greatest effect at ratios of R/t below 500. For values of the ratio above 500 the ratio of σ/σ_{c1} is observed to remain nearly constant. The logical reason for the above effect is that for larger, thinner shells with a higher ratio of R/t , there are likely to be more imperfections. There are more imperfections found due to the larger surface area.

7. Conclusions

This investigation adds to the general study on experimental effort developing general information on the buckling of thin-walled cylindrical shells. The polyester film, Mylar, is adaptable and definitely a significant aid in such a study. The ease in the fabrication of cylindrical shells out of Mylar sheet, and the repeatability phenomenon, enabling the model to be buckled several times at virtually the same buckling load, enables the investigator to obtain considerable test data with a minimum of experimental effort and time.

The investigation conducted confirms that the number of circumferential buckles, N , is a function of both the parameters, L/R and R/t . When R/t is held constant, N will decrease with increasing values of the parameter L/R . When L/R is held at a constant value, N will increase with increasing values of the ratio of R/t . It is important to determine the value of N because it is an index of the buckling mode, which in turn controls the behavior and strength of the shell after buckling. From the investigation it was found that N accurately follows the approximate theoretical expression:

$$N = 2.2 \left[\frac{R/t}{L/R} \right]^{1/4},$$

developed by deNeufville.

It is also concluded, for the range of the parameters, L/R and R/t , considered in the investigation, that the ratio of the buckling stress to that of the classical buckling stress is:

(1) independent of the parameter L/R , and (2) dependent on the parameter R/t , with increasing values of the ratio of R/t resulting in a decrease in σ/σ_{cl} . The parameter R/t has its greatest effect at ratios of R/t below 500. For values of the ratio above 500 the ratio of σ/σ_{cl} remains nearly constant.

Further experimental effort on the buckling of cylindrical shells using the polyester film, Mylar, can definitely expand and contribute greatly to the general understanding of the buckling

mechanism. Likely areas of investigation are: (1) larger spread in the variation of the parameter L/R , (2) values of the R/t parameter greater than 1600, (3) effect of reinforcement of the shell wall by the use of circumferential stiffeners, (4) effect of the axial overlap used for the bonding of the lap joint, used in this investigation and earlier investigations by Weingarten, Morgan, and Seide, and de Neufville, and (5) effects of various end fixity conditions (deNeufville stated that, from a theoretical point of view, the study of the buckling mechanism has no preferable form of edge restraint [7].)

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for elastic stability of thin shell structures.
Space Technology Laboratories Report STL/TR-
60-0000-19425, Los Angeles, California,
December, 1960.

Table I
Schedule of Parameters

t	R	L	R/t	L/R
7.5 mils	2"	1.83"	267	.918
↓	↓	2.5"	↓	1.25
	↓	4.0"	↓	2.0
	3"	2.75"	400	.918
	↓	3.75"	↓	1.25
	↓	6.0"	↓	2.0
	5"	4.58"	667	.918
	↓	6.25"	↓	1.25
	↓	10.0"	↓	2.0
10 mils	2"	1.83"	200	.918
↓	↓	2.5"	↓	1.25
	↓	4.0"	↓	2.0
	3"	2.75"	300	.918
	↓	3.75"	↓	1.25
	↓	6.0"	↓	2.0
	5"	4.58"	500	.918
	↓	6.25"	↓	1.25
	↓	10.0"	↓	2.0

Table II
Summary of Data for Control Tests

$R/t = 400$ $L/R = 1.58$
 $t = 7.5$ mils $R = 3.0$ inches

Model	Run No.	Buckling load (lbs.)	N
A ↓	1	74	9.0
	2	72	9.1
	3	70	8.9
	4	66	8.9
	5	66	8.7
	6	68	9.0
B ↓	1	70	9.1
	2	72	9.4
	3	72	9.4
	4	69	9.0
	5	71	9.0
	6	69	8.8
C ↓	1	72	9.0
	2	72	9.0
	3	70	8.6
	4	72	9.0
	5	74	8.9
	6	74	9.2
	7	72	9.2

Table II (Cont'd.)

Model	Run No.	Buckling load (lbs.)	N
D ↓	1	66	9.4
	2	68	8.9
	3	66	8.8
	4	62	9.5
	5	64	9.5
E ↓	1	72	9.0
	2	74	9.3
	3	75	9.3
	4	78	9.4
	5	78	9.0
	6	78	8.9

Table III
Summary of Data for Test Models

* reject						
R/t	L/R	t (mils)	Model No.	Average σ / σ_{cl}	No. of tests	Average N
400 ↓ ↓ ↓ ↓ ↓ ↓ ↓ ↓	.916 ↓	7.5 ↓ ↓ ↓ ↓ ↓ ↓ ↓	7	.404	6	10.5
			8	.399	6	10.6
			9	.403	5	10.3
	1.25 ↓		1	.448	9	9.6
			2	.458	10	9.5
			3	.518	8	9.5
			3A	.417	7	9.5
	2.0 ↓		13*	.588	7	8.4
			14	.407	7	8.0
			15	.362	6	8.2
300 ↓ ↓ ↓ ↓ ↓ ↓ ↓	.916 ↓	10.0 ↓ ↓ ↓ ↓ ↓ ↓ ↓	10	.448	8	9.2
			11	poor fabrication reject		
			12	.378	6	9.4
	1.25 ↓		4	.457	6	8.5
			5	.505	5	8.5
			6	.454	7	8.8
	2.0 ↓		16	.374	9	8.1
			17	.441	7	8.0
			18	.553	8	7.9
267 ↓ ↓ ↓	.916 ↓	7.5 ↓ ↓ ↓	26	.521	5	9.2
			27*	.653	6	9.1
			28	.465	5	9.1
	1.25 ↓		22	.575	10	8.4

Table III (Cont'd.)

R/t	L/R	t (mils)	Model No.	Average σ/σ_{cl}	No. of tests	Average N
200	↓ 2.0 ↓ .916 ↓ 1.25 ↓ 2.0 ↓ .916 ↓ 1.25 ↓ 2.0 ↓	10.0	23	.598	6	8.5
			24	.473	6	8.4
			25	.526	7	8.4
			19*	.655	9	7.4
			20	.563	7	7.5
			21	.605	8	7.4
			36	.498	7	8.4
			37*	.388	5	8.3
			38	.505	7	8.5
			32	.534	6	7.9
667	↓ 1.25 ↓ 2.0 ↓ .916 ↓ 1.25 ↓ 2.0 ↓	7.5	33	.493	7	7.9
			34	.661	6	7.8
			35	.622	6	7.8
			29	poor fabrication reject		
			30	.589	8	6.9
			31	.486	6	6.9
			43	.417	5	11.2
			44	.368	5	11.1
			41	.354	6	10.5
			42	.373	5	10.5
	↓ 2.0 ↓		39	.417	5	9.7
			40	.410	6	9.5

Table III (Cont'd.)

R/t	L/R	t (mils)	Model No.	Average σ/σ_{cl}	No. of tests	Average N
500	.916	10.0	50	.385	6	10.4
↓	↓	↓	51	.398	5	10.6
	1.25		47	.351	6	9.7
	↓		48	.432	6	9.8
	↓		49	.368	5	9.6
	2.0		45	.384	6	8.8
↓	↓	↓	46	.382	6	8.8

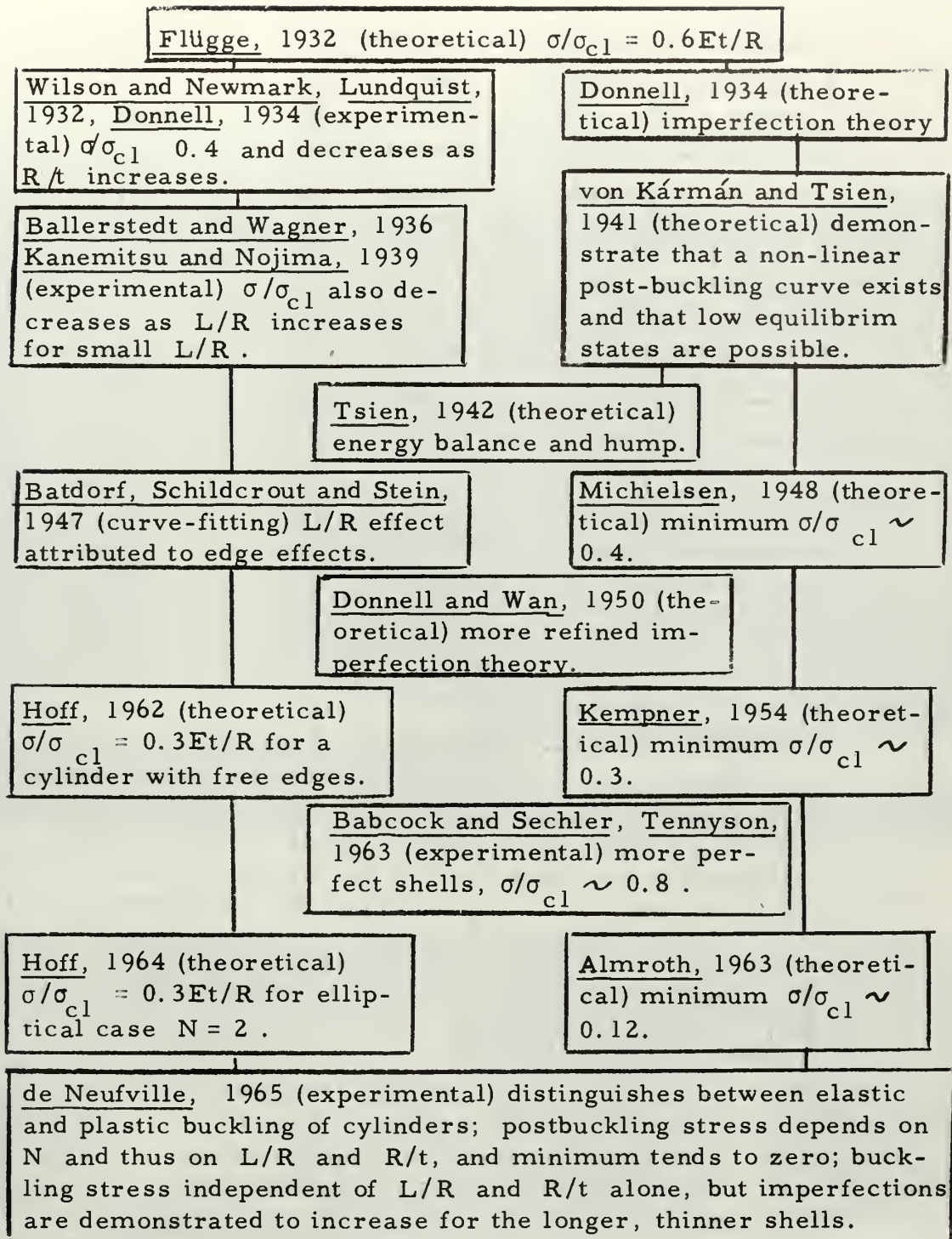


Figure 1.
Chronology of investigation of buckling stress
of thin cylinders [10].

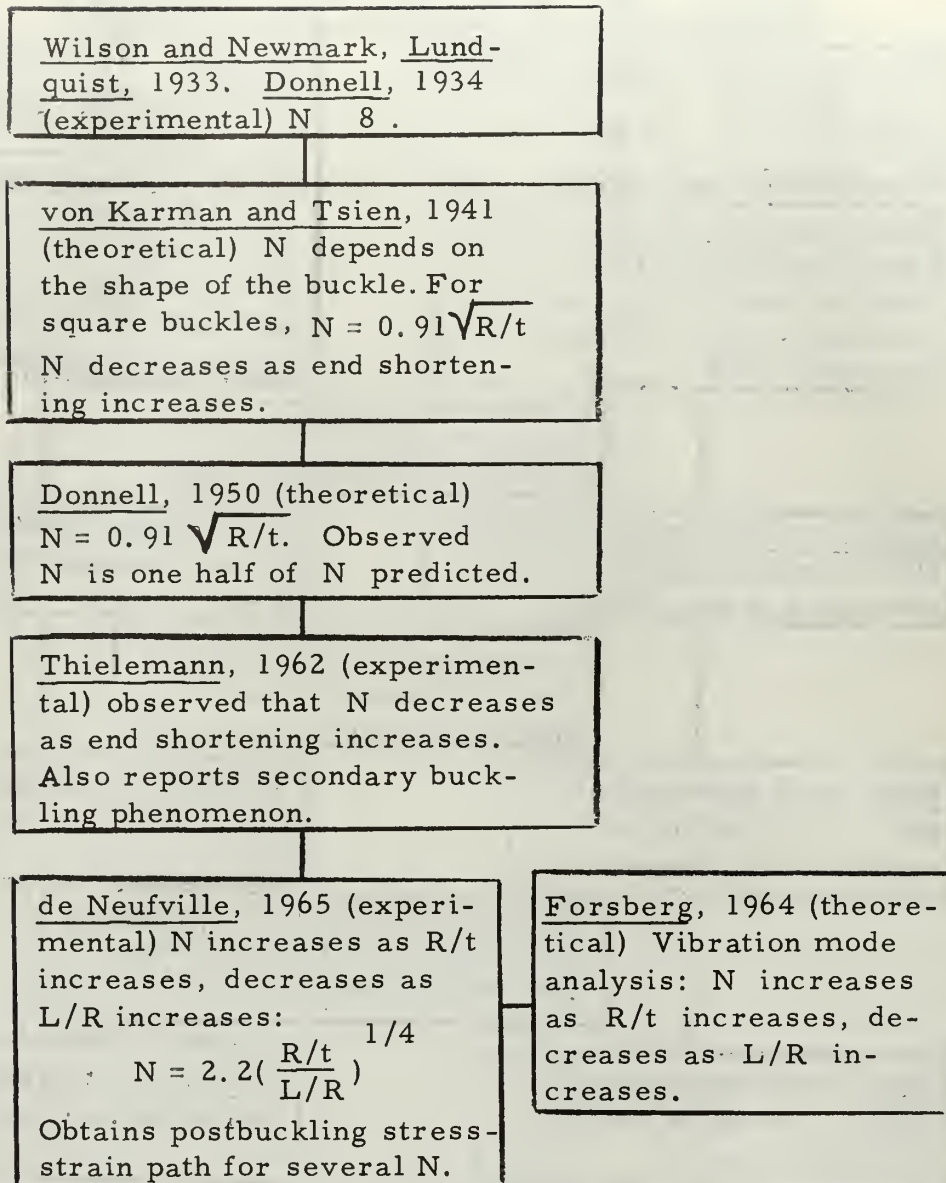
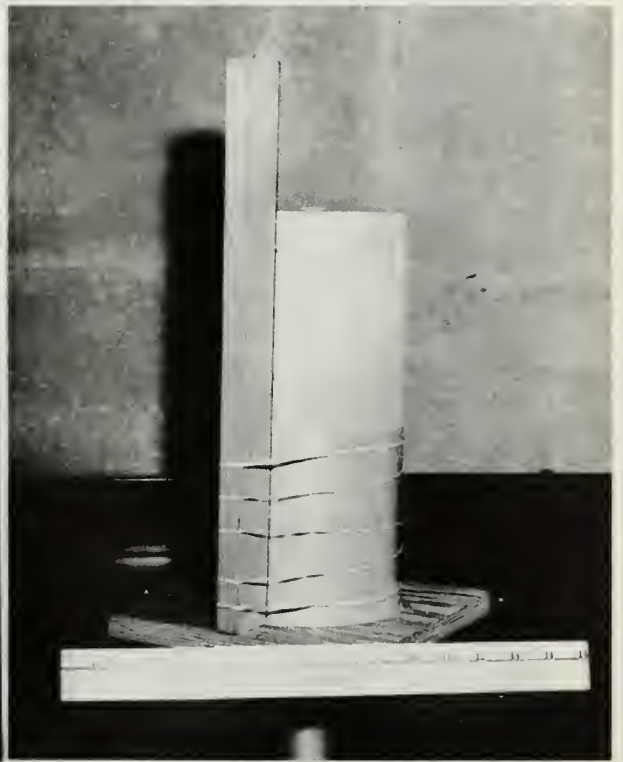


Figure 2.
Chronology of the investigation of the buckling mode of thin cylinders [10].



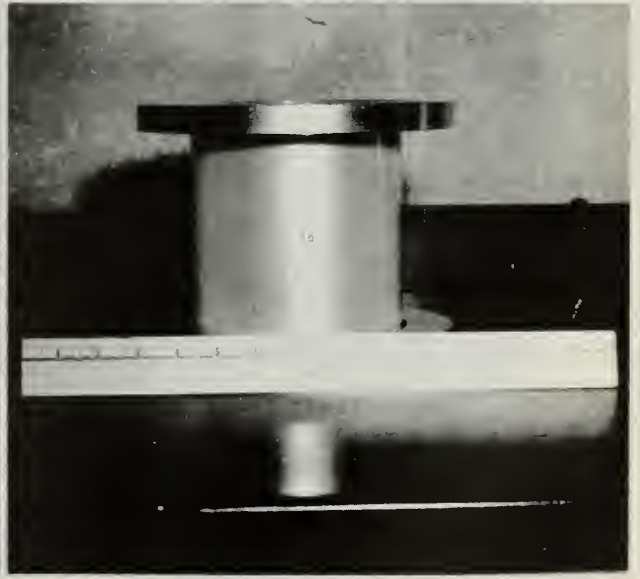
a



c



b



d

Figure 3. Test model fabrication procedure



Figure 4. View of the various test model sizes investigated

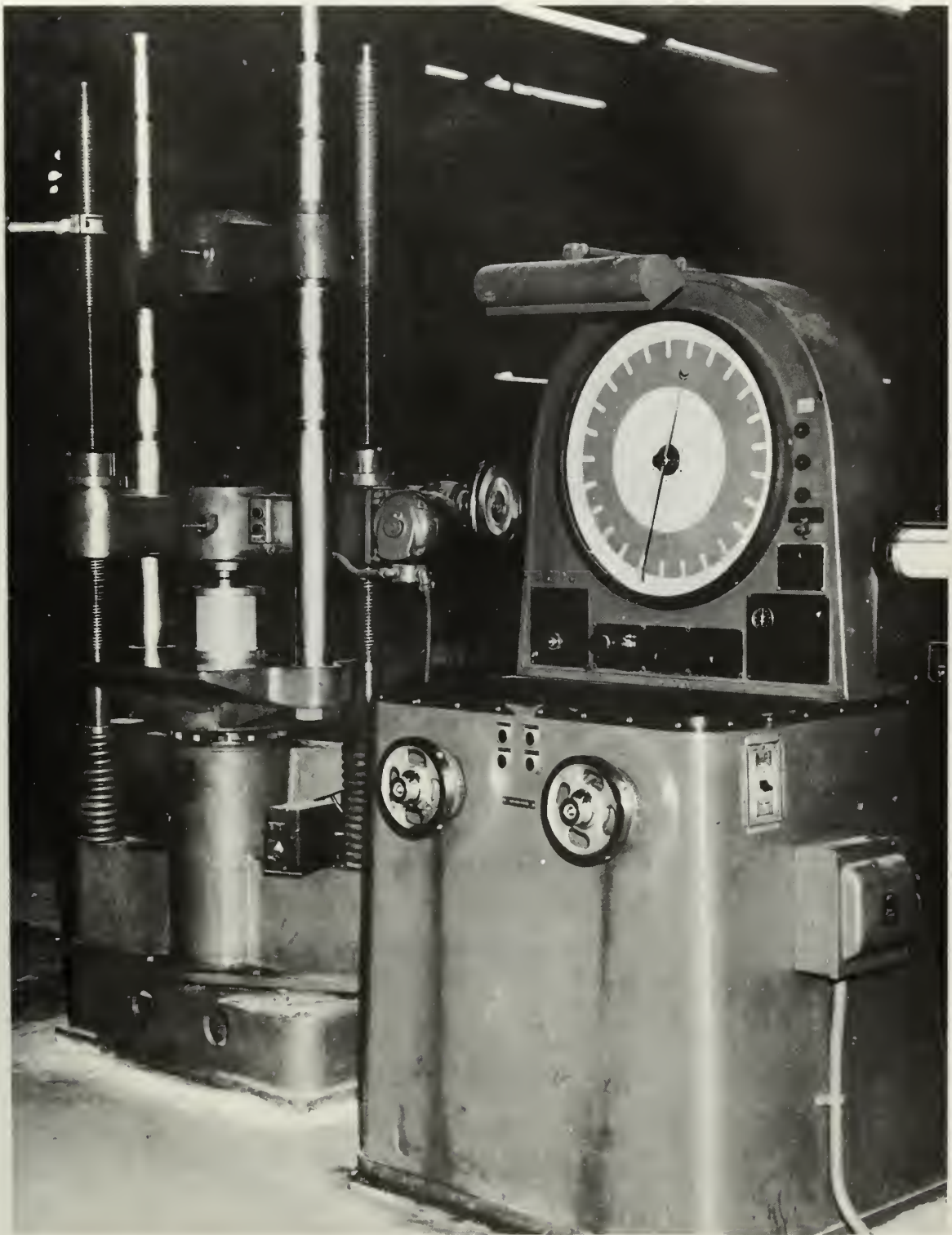


Figure 5. Testing machine and model setup



Figure 6. Photograph of a typical buckling pattern observed

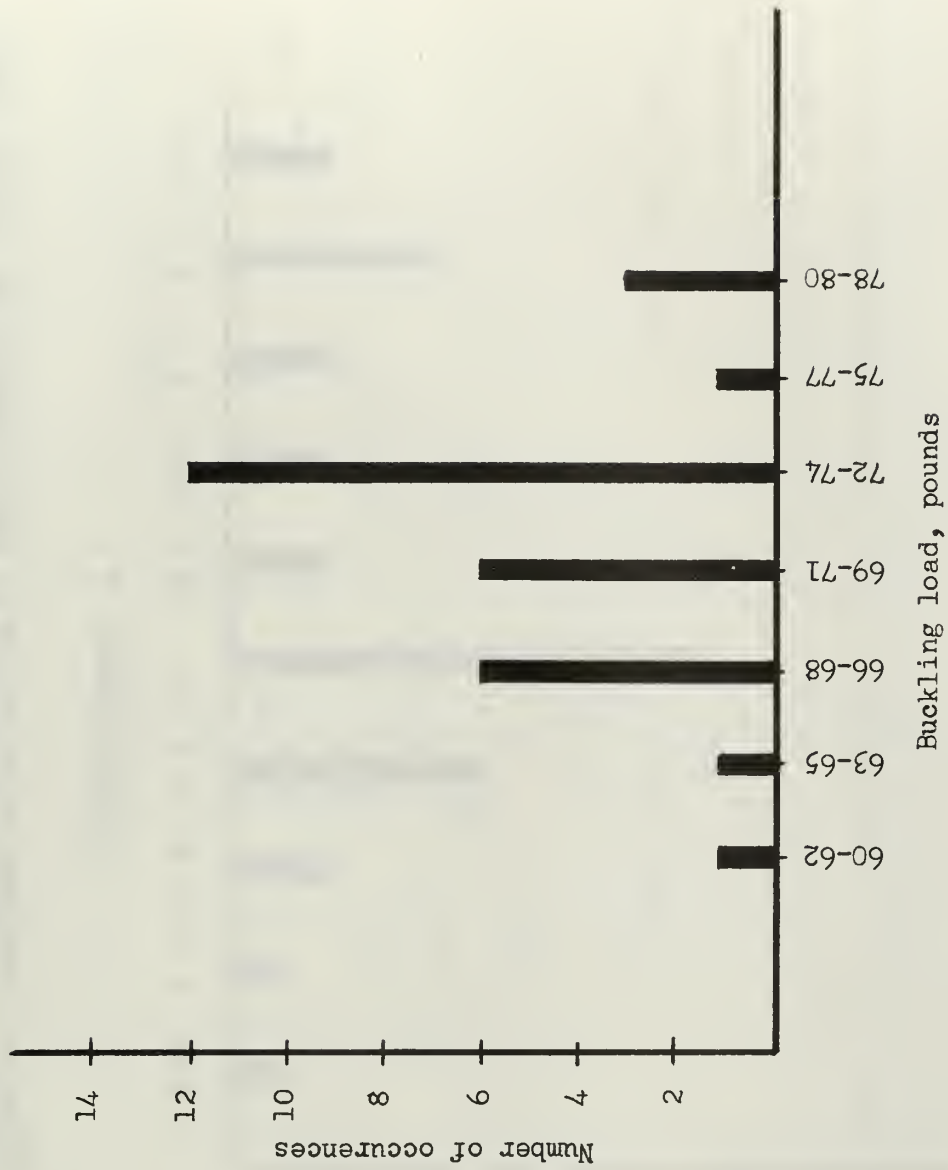


Figure 7. Frequency distribution of buckling load for control tests

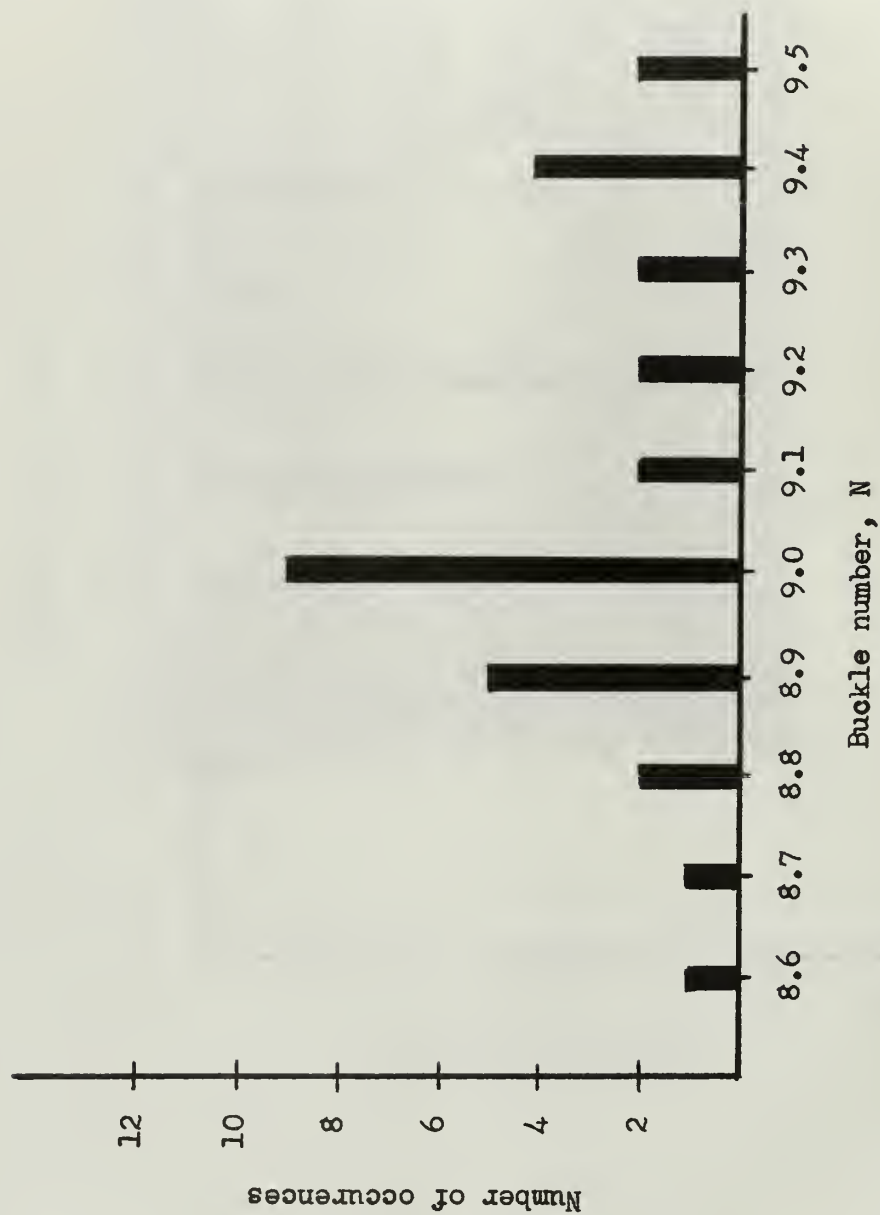


Figure 8. Frequency distribution of buckle number for control tests

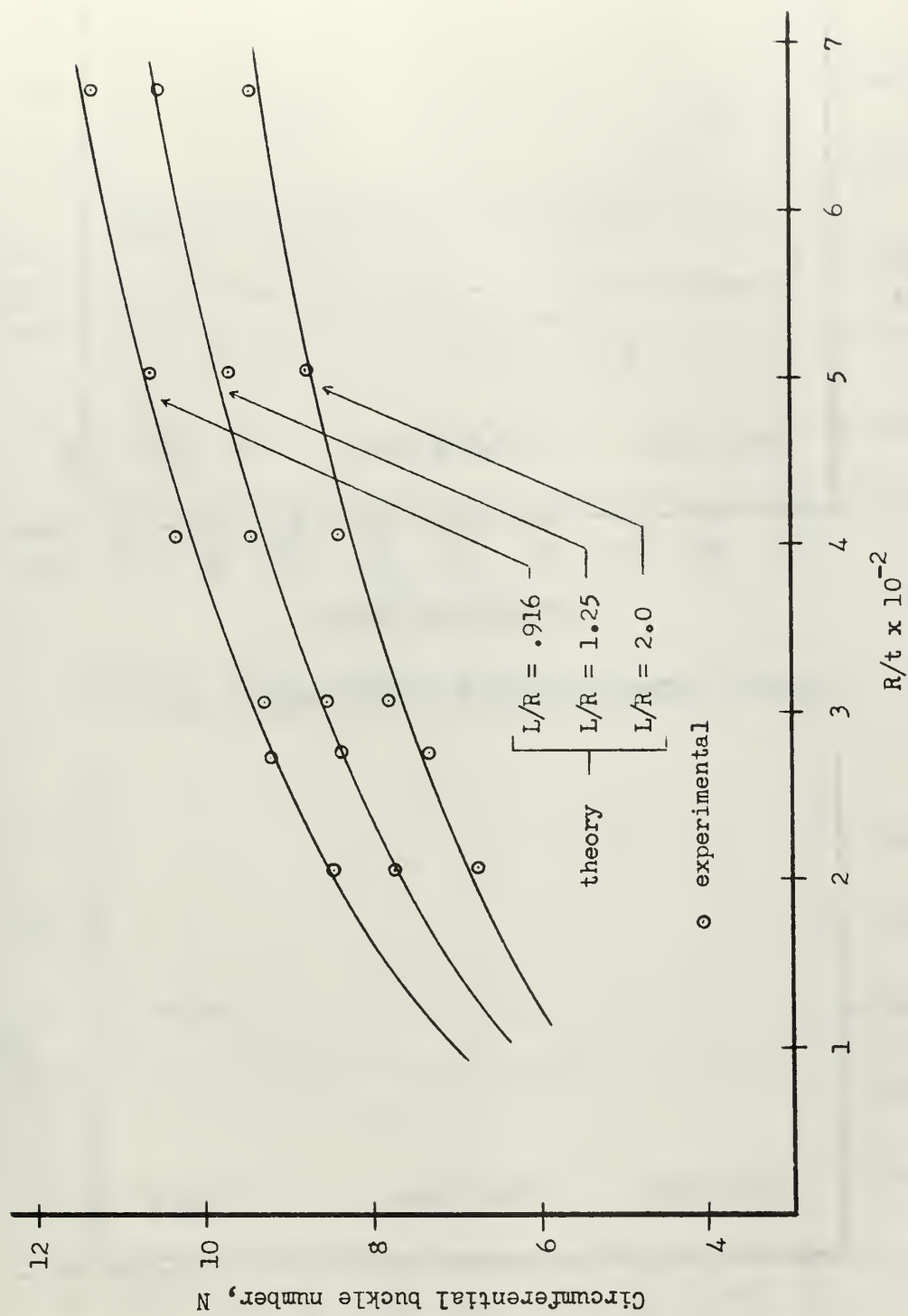


Figure 9. Variation of N with R/t and L/R , theory and experimental results compared

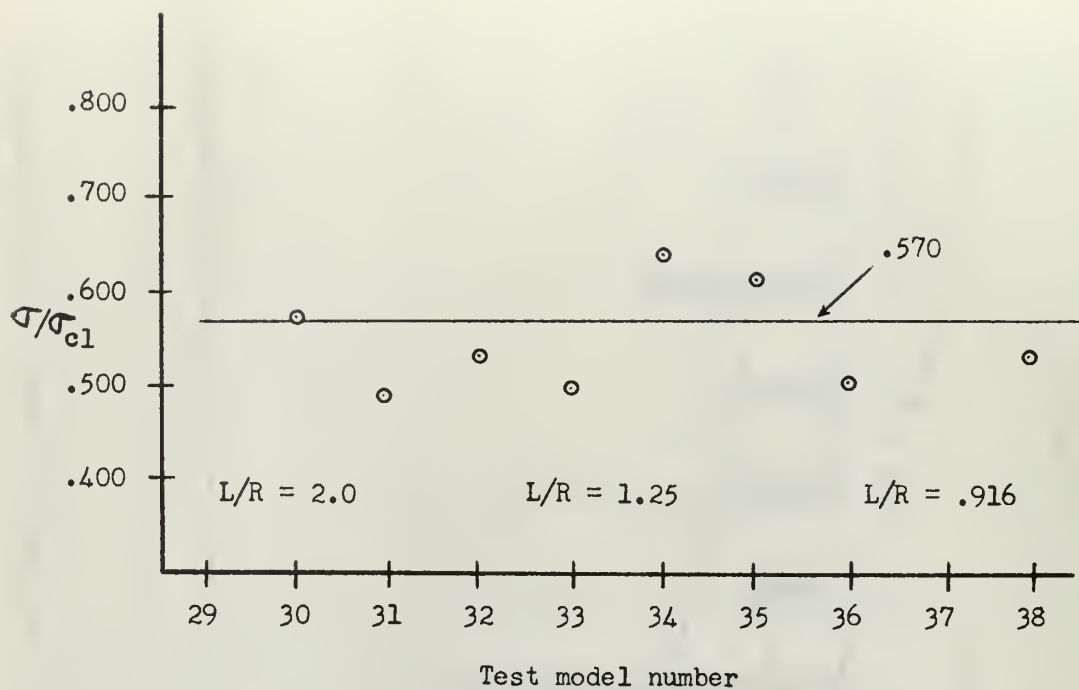


Figure 10. σ/σ_{c1} versus L/R for R/t equal to 200

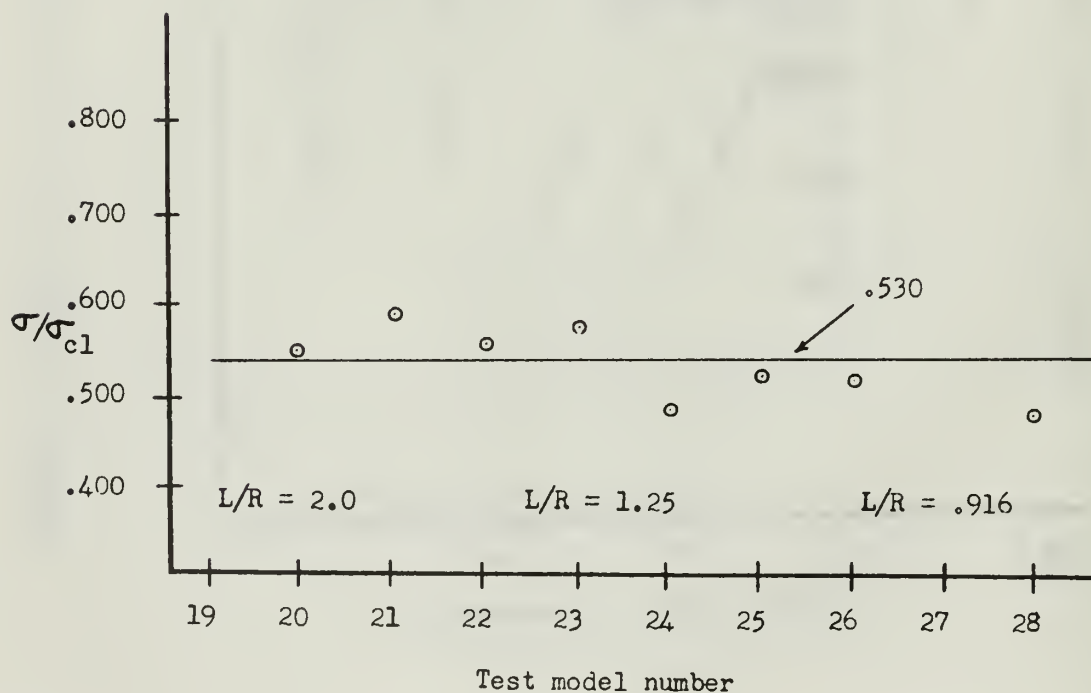


Figure 11. σ/σ_{c1} versus L/R for R/t equal to 267

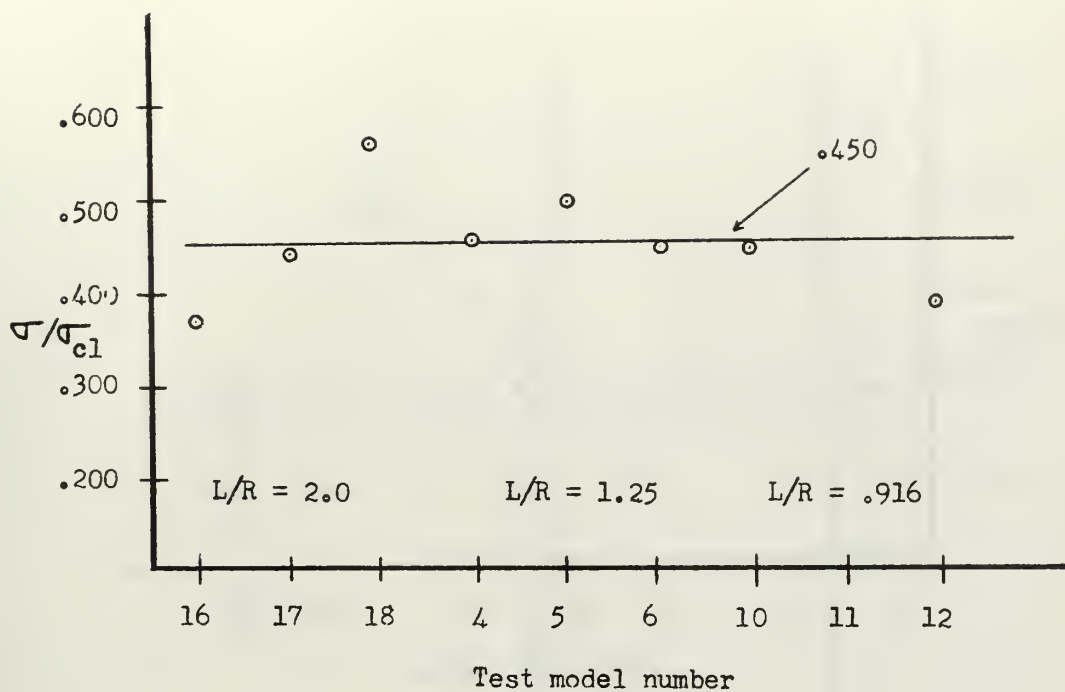


Figure 12. σ/σ_{c1} versus L/R for R/t equal to 300

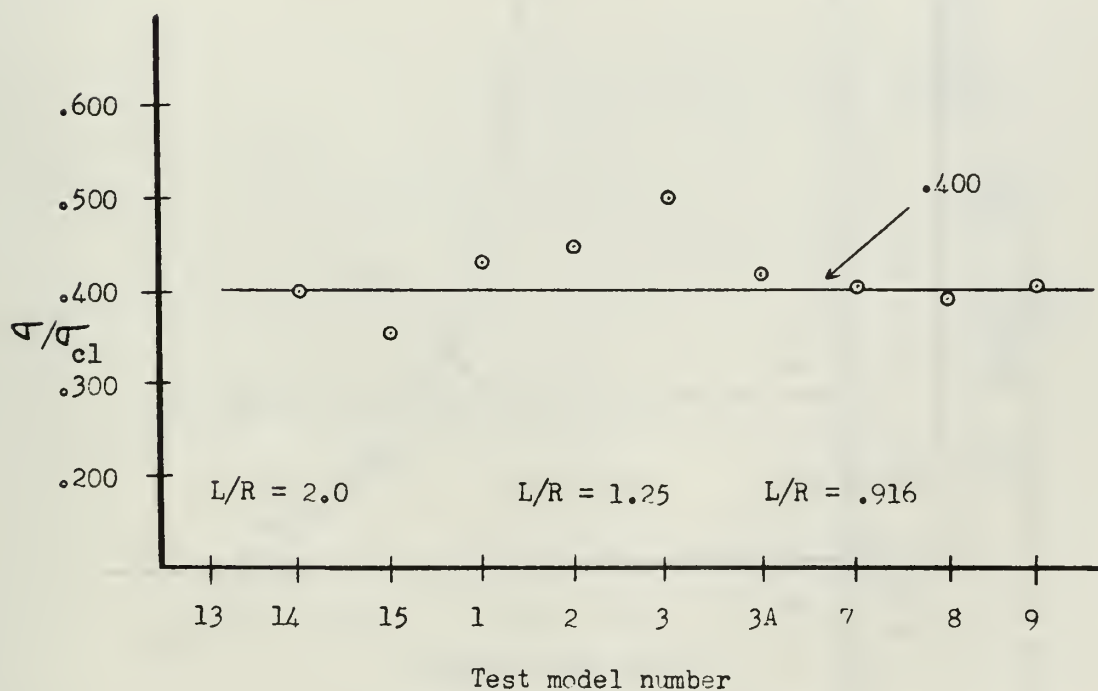


Figure 13. σ/σ_{c1} versus L/R for R/t equal to 400

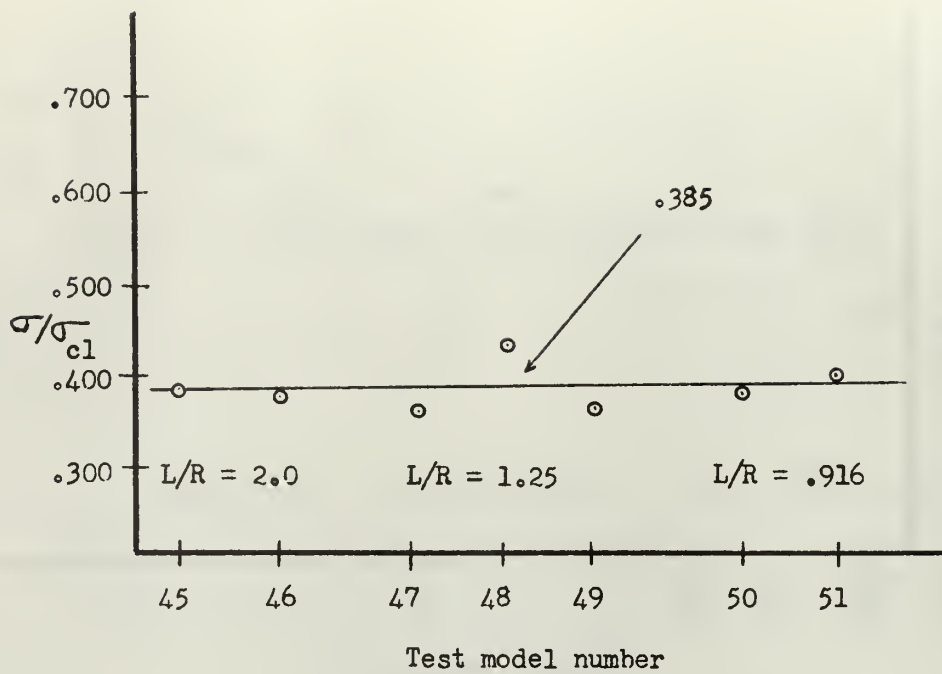


Figure 14. σ/σ_{c1} versus L/R for R/t equal to 500

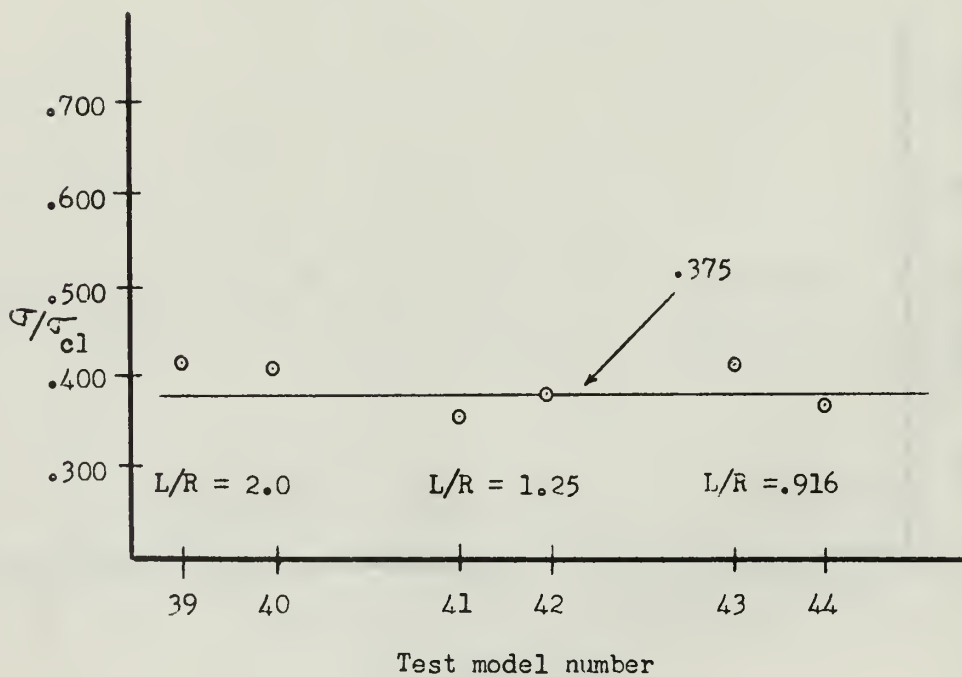


Figure 15. σ/σ_{c1} versus L/R for R/t equal to 667

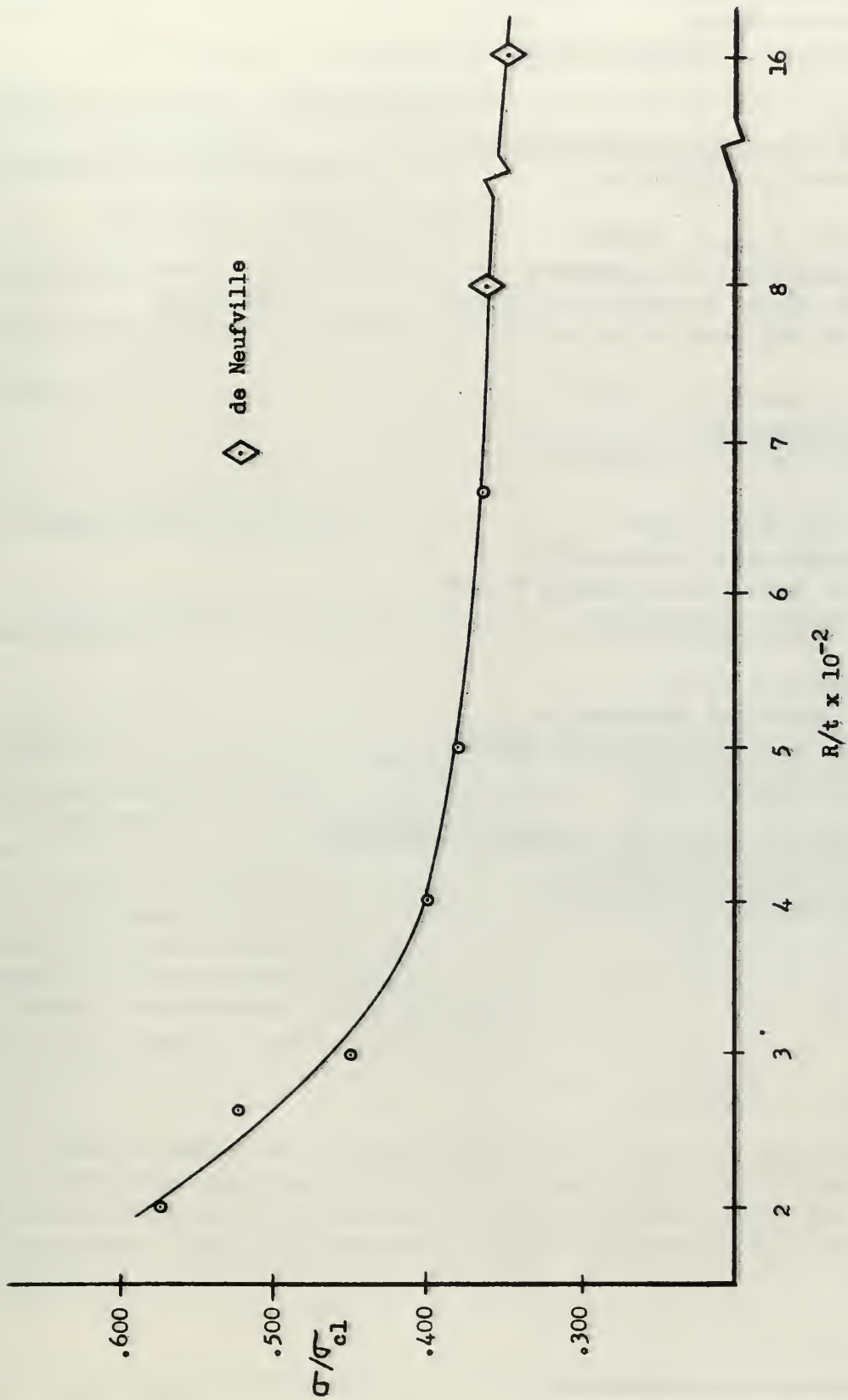


Figure 16. Variation of σ/σ_{c1} with R/t

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13. ABSTRACT The objective of research in the field of axially compressed cylinders is the establishment of sound design principles. Only through additional analytical and experimental effort can this goal be met. The most critical need at the present is believed to be for additional tests, which, unlike most previous tests, will be designed specifically to provide general information about the buckling process. This investigation adds to the general study on the buckling of thin-walled cylindrical shells fabricated from the polyester film, Mylar. It was concluded that the number of circumferential buckles, which controls the postbuckling behavior of cylinders, is dependent on the geometric parameters, R/t and L/R . Such dependence is approximated by the relation: $N = 2.2 \left[\frac{R/t}{L/R} \right]^{1/4}$ It is also concluded, for the range of the R/t and L/R parameters considered in this investigation, that the ratio of the buckling stress to that of the classical buckling stress is: (1) independent of the parameter L/R , and (2) dependent on the parameter R/t , with increasing values of the ratio R/t causing a decrease in σ/σ_{cl} .			

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Buckling						
Cylinders						
Mylar						

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